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# INFLUENCE OF STRESSES IN A MODIFIED NON-METALLIC SPUR GEAR PAIR

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***ABSTRACT***

***A gear is a rotating machine part having cut teeth, or cogs, which mesh with another toothed part in order to transmit torque. In order to avoid undercutting and interference, addendum modification of the gear tooth is carried out in this paper, a standard and a profile corrected carried out spur gear pair is modeled. The material used is plastic (Nylon-66). The gear ratio assumed was 1.5 with a module of 10 and correction factor taken to be 0.5. The dimension of the models was arrived at by theoretical calculations. In this paper, a model part is done with PRO-E for the modeling of the spur gear for both standard and profile corrected tooth. The finite element model of the gear tooth is imported to an analysis software ANSYS to study the bending effects for different modules. Results were obtained from the comparisons made for the bending stress variations for both the standard plastic spur gear tooth and the profile corrected plastic spur gear. The results of the study for the addendum modified tooth showed a decrease in bending stresses for the wider tooth for the same loading.***

***KEYWORDS:*** *Spur Gear, Bending, Contact & Stress Analysis*

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**Original Article**

## INTRODUCTION

Gears operate in pairs, when pinion is the driver, it results in step down drive in which the output speed decreases and the torque increases. On the other hand, when the gear is the driver, it results in step up drive in which, the output speed increases and the torque decreases. Much research had been carried out from the yester years till today on gears. There were varied difference of opinion regarding gear tooth failures, and there still exists a need for much more research on the behavior of surfaces in a pair of gear in contact. In the year 1950, Dr. W. A . Tuplin (7) concentrated on bending stress. Bending stress was a possible factor in failure of a gear. If the maximum stress in a gear tooth is less than the fatigue limit for the material, the tooth should not fail even after prolonged running. In 1957 Niemann and Rettig (5) tested a number of steel gears. The deflection of the tip was measured under both static and dynamic condition. In 1953, Strauch (6) in his paper presented that there was continuous error due to vibrations in uncorrected gears i.e. winter tested steel gears. The results of the bending stresses were obtained by the assumption of load application on the tooth tip teeth. In 1893, Lewis (9) presented in his paper, the Lewis formula for bending stress. J. D. Andrews (14) in his paper investigated the fillet stresses predicted in both external and internal forms of spur gear teeth using Finite Element Method. Sorincananau (17) in

his paper investigated 2D versus 3D analysis for stresses in the root region of the teeth. M. Beghini (18) in their paper proposed a simple method to reduce the transmission error for a given spur gear by means of profile modification parameters. Ravichandra patchigolla et.al in his paper emphasizes the results of Finite element analysis and the effect of rim thickness on gear tooth bending stress. C. V. Spitas & Spitas (19) in their paper made a comparison of the bending strength between the circular fillet and the trochoidal fillet, in which the use of trochoidal fillet proved to be advantageous. Ivana Atanasovsk (20) discusses in detail using Finite element method models of involute spur gears for monitoring the stiffness and the base pitch deviation influence on the load distribution and the load capacity. Dr. I. G. H Van Melick (22) in his paper investigates steel and plastic gear transmission using numerical finite element and analytical methods to study the influence of stiffness of the gear material on the bending of the gear teeth. The change in the load sharing also changes the stresses. The Costopoulous et.al (23) proposed several tooth alternative design for increasing the load capacity. Buljanovic. K et.al (24) presented linear tip profile modification and compared the gear tooth root stresses of the profile modified one, with a standard spur gear tooth. Ali Raad Hassan (25) had presented in his paper, the results of contact analysis between two different spur gear teeth considered in different contact position. The results were compared theoretically. Dr. Eng. Ulrich Kissling's (26) approach uses an algorithm that includes conventional method for calculating tooth stiffness in regards to bending stress and shearing deformation. The results were then compared with FEM. He pointed out that the profile correction has an influence in reducing transmission error. Shanmugasundaram Shankar et.al (27) describes in their paper, the effect of profile modification in the root fillet region, with the use of a circular fillet for increase of the tooth strength of a spur gear using CAD.

## DESIGN

Gear Parameters used

**Gear Type:** Standard involute full - Depth Teeth.

**Input Parameters for Standard Spur Gear Pair**

**For Gear ratio 'i' = 1:1.5**

**Module:** 'm'= 10,

**Pressure Angle:** 'α' = $20^0$,

**No. of Teeth on Pinion:** '$Z_1$'= 14,

**No. of Teeth on Gear:** $Z_2$ = 21,

**Speed:** $n_1$=1000 (assumed),

a=175 mm

$n_2$=666.67 rpm

**Circular Pitch:** 'p'=31.415 mm

**Pitch Circle Diameter 'd'**

**Pinion:** '$d_1$' = 140 mm

**Gear:** '$d_2$'= 210 mm

**Thickness of the Teeth:** ‘S’ = 15.70 mm

**Outer Circle Diameter ‘$d_a$’**

**Pinion:** 160 mm

**Gear:** 230 mm

**Base Circle Diameter of the Gear ‘$d_b$’**

**Pinion:** ‘$d_{b1}$’=131.55 mm

**Gear:** ‘$d_{b2}$’ = 197.33 mm

**Root Circle Diameter ‘$d_{f1}$’**

**Pinion:** ‘$d_{f1}$’=116.86 mm

**Gear:** ‘$d_{f1}$’=186.86 mm

**Face Width:** ‘b’= 94.24 mm

**Input Parameters for Corrected Spur Gear Pair**

**For Gear Ratio ‘i’ = 1:1.5**

**Module:** ‘m’=10

**Pressure Angle:** ‘α’ = $20^0$

**No. of Teeth on Pinion:** ‘$z_1$’= 14

**No. of Teeth on Gear:** ‘$z_{2\,=}$ 21

**Correction Factor for Pinion:** ‘$X_1$’ = 0.5

**Correction Factor for Gear:** ‘$X_2$’ = -0.5

**Speed:** n1=1000 (assumed)

**Speed Ratio:** $'I' = \frac{1000}{n2} = \frac{21}{14}$

$n_2$=666.67 rpm

**Circular Pitch:** ‘p’=31.415 mm

**Pitch Circle Diamete r‘d’**

**Pinion:** ‘$d_1$’**=**140 mm

**Gear:** ‘$d_2$’ =210 mm

**Thickness of the Teeth ‘S’**

**Pinion:** ‘$S_1$’=19.34 mm

**Gear:** ‘$S_2$’**=**12.06 mm

**Outside Diameter of the Gear '$d_a$'**

**Pinion:** '$d_{a1}$'=170 mm

**Gear:** '$d_{a2}$'=220 mm

**Base Circle Diameter of the Gear '$d_b$'**

**Pinion:** '$d_{b1}$' = 140 $\cos 20^0$=131.55 mm

**Gear:** '$d_{b2}$' = 210 $\cos 20^0$=197.335 mm

**Root Diameter of Gear '$d_r$'**

**Pinion:** '$d_{r1}$'=125 mm

**Gear:** '$d_{r2}$'=175 mm

**Face Width:** 'b' =94.247 mm

**Total Depth:** 'h'=22.5 mm

## STRESSES AND LOAD CALCULATION

Modulus of elasticity (E)=5.5x103 N/mm$^2$

Poisson's ratio $\mu$=0.420

Yield stress $\sigma_y$=110 N/mm$^2$

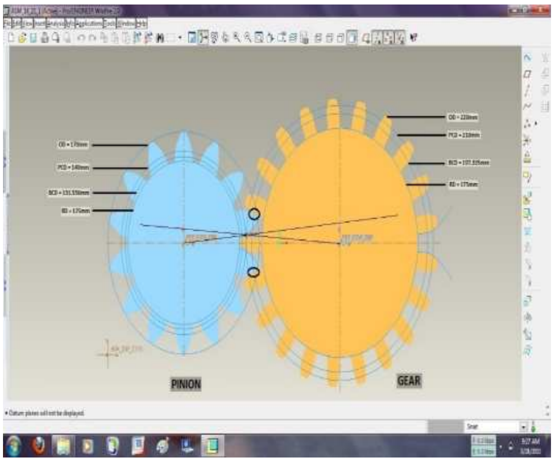



**Figure 1: Meshing Model of a Corrected Spur Gear Pair at $0^0$ with Case 4 & 5 Point of Contact**

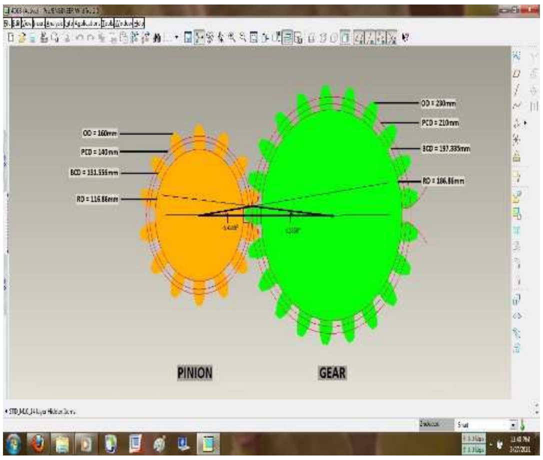



**Figure 2: Meshed Model of Standard Spur Gear Pair**

BHN=<350

Tensile strength $\sigma_u$=170 N/mm$^2$

Design bending stress $\sigma_b$=85 N/mm$^2$

Induced bending stress $\sigma_b$=8.07*10$^3$ N/mm$^2$

Design contact stress $\sigma_c$=180 N/mm$^2$

Induced contact stress $\sigma_c$=48.067 N/mm$^2$

Tangential Load ($F_t$)=9.553 N

Normal Load ($F_n$)=10.144 N

Radial Load ($F_r$)=3.469 N

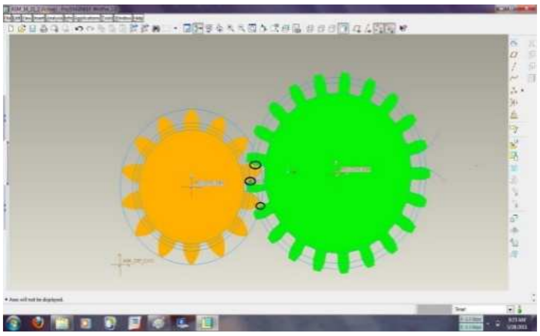


**Figure 3: Meshed Model of Corrected Spur Gear with 1, 2 & 3 Point of Contact**

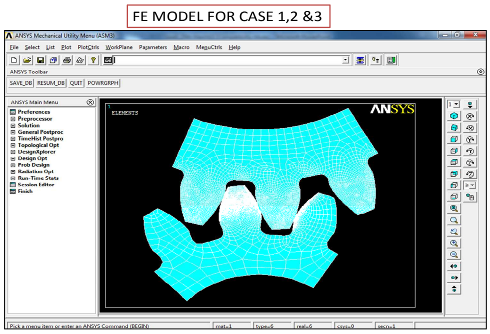



**Figure 4: FEA Model for Case 1, 2 & 3**

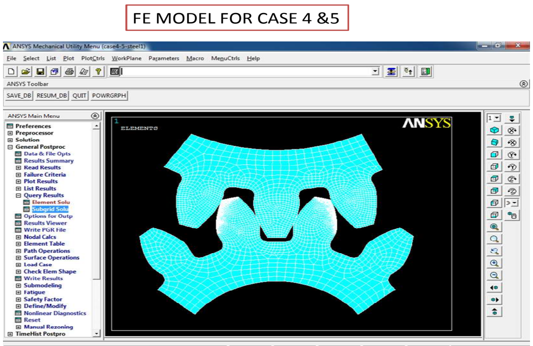



**Figure 5: FEA Model for Case 4 & 5**

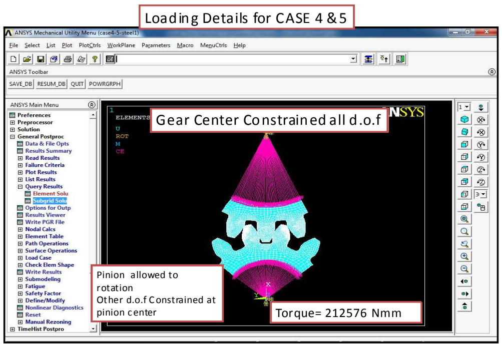



**Figure 6: Boundary Condition for Case 1, 2 & 3**

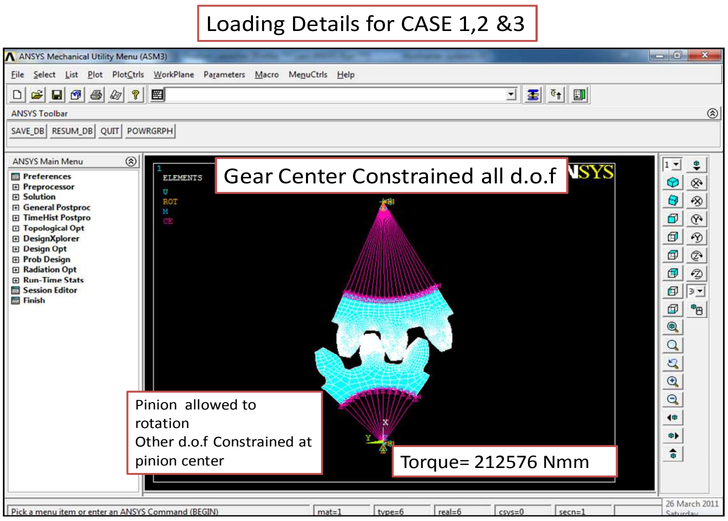



**Figure 7: Boundary Condition for Case 1, 2 & 3**

### Standard Gear Pair Bending Stress & Contact Stress Analysis

Bending Stress–13N/mm2 Nylon- case1

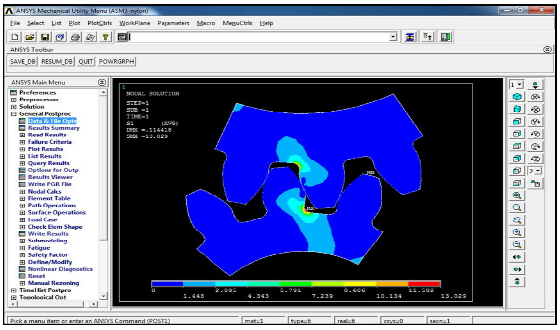


**Figure 8: Bending Stress Case 1**

FE MODEL FOR CASE 4 &5

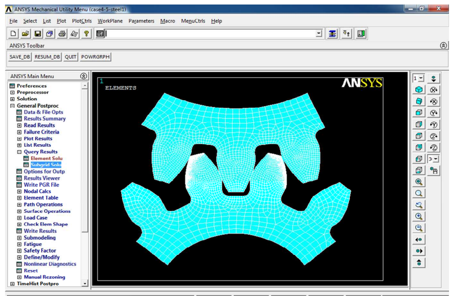


**Figure 9: Contact Stress Case 1**

**Contact Pressure – 36.4N/mm²** **Nylon- case 2**

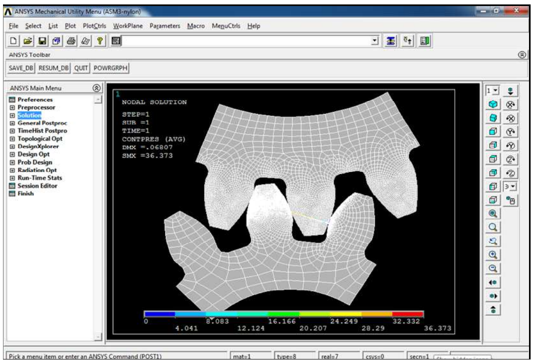


**Figure 10: Contact Stress Case 2**

Bending Stress–18.6N/mm2 Nylon- case3

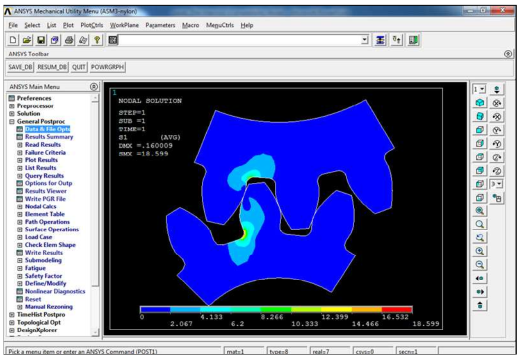


**Figure 11: Bending Stress Case 3**

Contact Pressure – 35.89 N/mm2 Nylon- case3

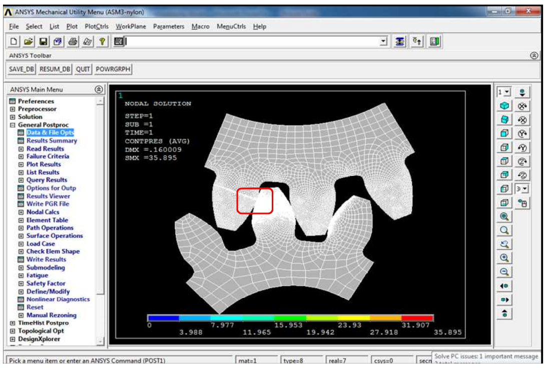


**Figure 12: Contact Stress Case 3**

Bending Stress –23.49N/mm² Nylon- case4

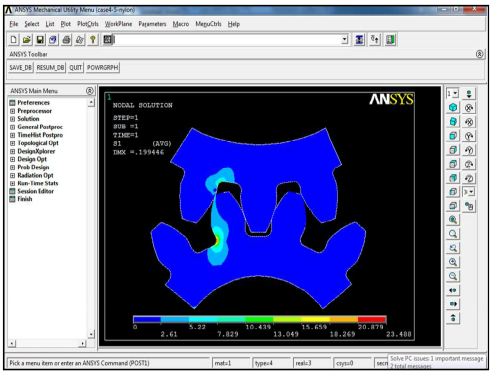


**Figure 13: Bending Stress Case 4**

Contact Pressure–45.875N/mm2 Nylon- case4

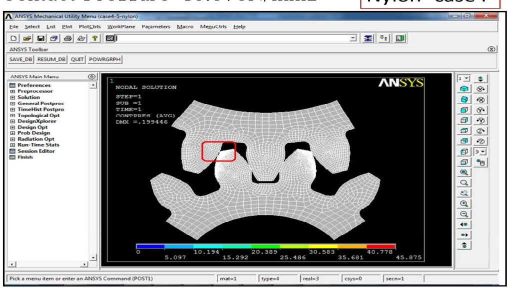


**Figure 14: Contact Stress Case 4**

## Corrected Gear Pair Bending Stress & Contact Stress Analysis

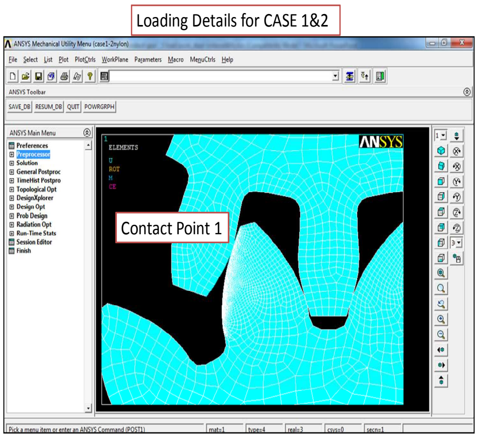



**Figure 15: Contact Point 1**

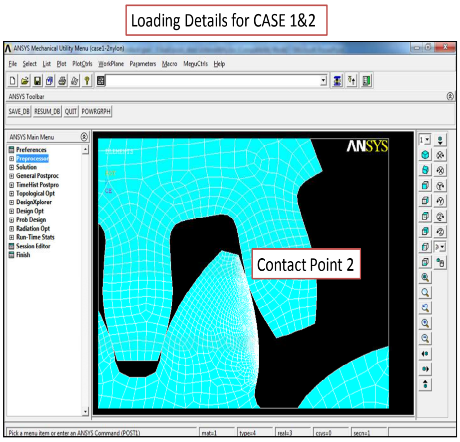
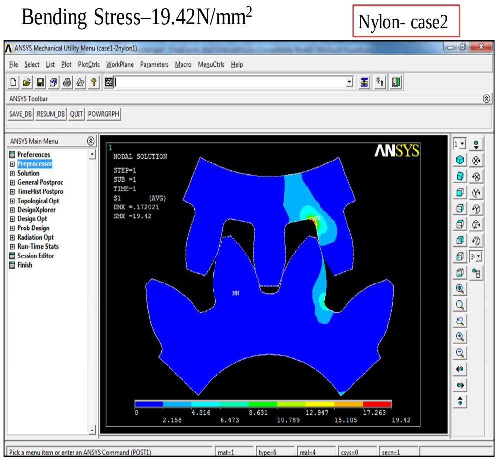



**Figure 16: Contact Point 2**



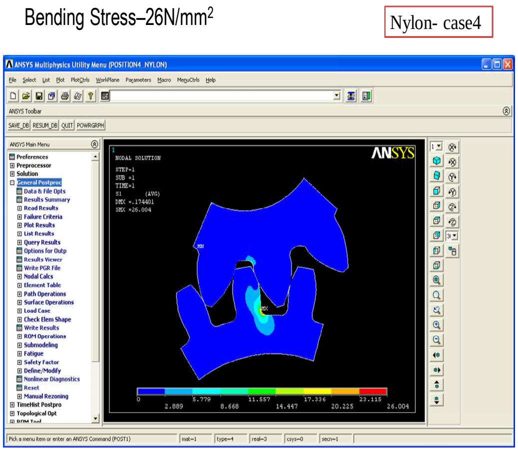
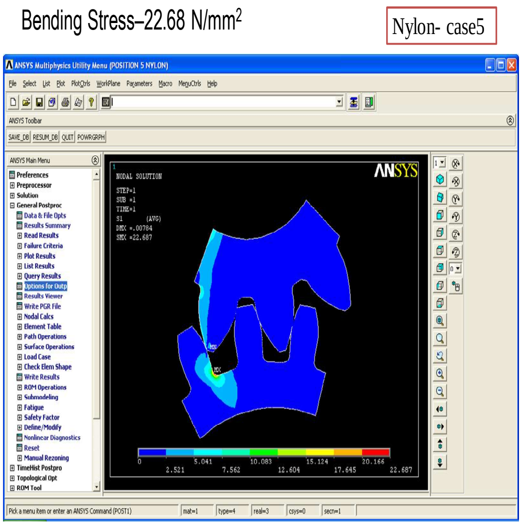




**Figure 17: Bending Stress Case 1 & Case 2**



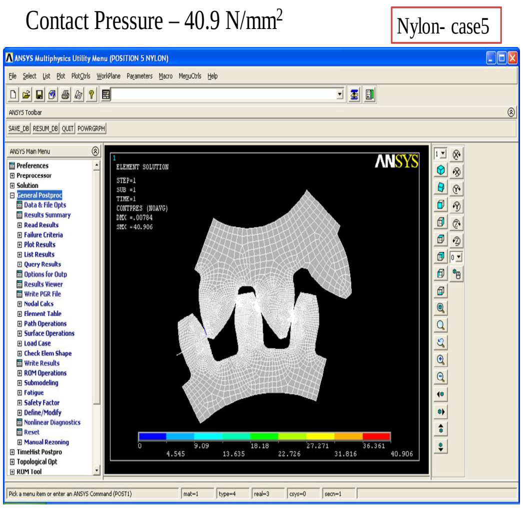




**Figure 18: Bending Stress & Contact Stress Case 5**

## ANALYTICAL RESULT AND GRAPHICAL ILLUSTRATION

**Table 1**

| Point of Contact | Plastic Material | Contact Stress $N/mm^2$ | Bending Stress $N/mm^2$ |
|---|---|---|---|
| Case 1 | Standard | 49 | 13 |
| | Corrected | 44.45 | 17.75 |
| Case 2 | Standard | 36.4 | 14.75 |
| | Corrected | 43.86 | 19.42 |
| Case 3 | Standard | 35.89 | 18.6 |
| | Corrected | 43.12 | 20.54 |
| Case 4 | Standard | 45.875 | 23.49 |
| | Corrected | 41.23 | 26 |
| Case 5 | Standard | 43.76 | 23.36 |
| | Corrected | 40.9 | 22.68 |

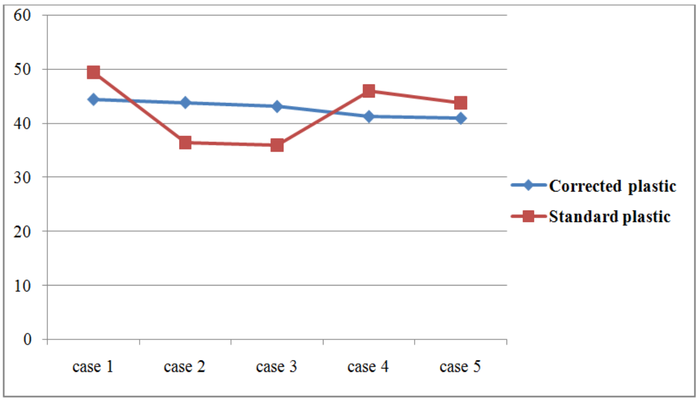



**Figure 19: Comparison of Contact Stress Results between Standard Plastic and Corrected Plastic Gear Pair**

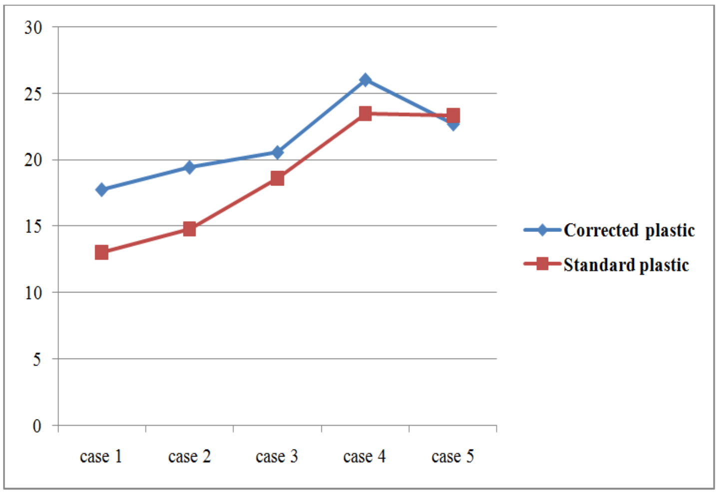



**Figure 20: Comparison of Bending Stress Results between Standard Plastic and Corrected Plastic Gear Pair**

## CONCLUSIONS

In this paper, the stress analysis of standard and profile corrected plastic gear pair was done. The geometry of the gear was modeled in modeling software Pro-E. The meshing was carried out by HYPERMESH software, and the analysis

was completed using ANSYS10 software. It was seen from the results that the theoretical value of the bending stress does not agree with the analytical result for a standard gear. It was found to be high, but was within the permissible bending stress limit for a plastic material. Hence, it is agreed that the profile corrected results obtained are also true. The Bending stresses for a standard and profile corrected tooth was found to be high. This is due to the fact of the material taken into consideration. Though the bending stress depends upon geometry, but due to load sharing, deflection is more for plastic material, it has a tendency to bend more for the calculated torque. The torque can thus be revised for suitable application. The maximum contact stress occurs in case 4, which is the pitch point. Here, the influence of young's modulus plays a role. Under static contact analysis, the contact stresses were found to be within limits calculated theoretically, but the results may vary severely for dynamic loading.

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